Design and Development of a Power Takeoff Shaft

Domenick J. Gargiulo*
Grumman Aerospace Corporation, Bethpage, New York

The X-29 forward-swept wing aircraft required a power takeoff shaft to transmit power from the starter to the engine during engine starting and to transmit power from the engine to the remote gearbox during ground and flight operations of aircraft accessories. Shaft design requirements of high speed and great length could not be met with off-the-shelf designs. Meeting adequate critical speed margin posed a significant design problem. Various shaft concepts were studied. The final selection was a graphite/epoxy center shaft utilizing metal flexible end assemblies. Extensive testing at unit, subsystem, and system levels proved the adequacy of the design.

Introduction

THIS paper details the design and development efforts that were performed to provide a power takeoff (PTO) shaft for the X-29A aircraft.

The forward-swept wing (FSW) program was structured on the utilization of existing components wherever possible to save development time and money. As a result, the components selected were not specifically designed to be utilized together. For instance, the F-18 F404 engine used with the F-16 remote gearbox created the requirement for a new PTO shaft. Neither the F-16 nor F-18 shafts could be used due to length requirements and different end configurations. The F404 engine drive flange for the PTO shaft is set back from the engine's forward flange about 18 in. and the F-16 aircraft mounted accessory drive (AMAD) has rearward-facing accessories. This combination dictates an unusually long PTO shaft (31.5 in.). A general arrangement is presented in Figs. 1 and 2. This unusually long shaft, combined with its high operating speed (16,810 rpm at 100% speed), generated concern that an adequate critical speed margin would not be obtained. The critical speed is defined as the staff's speed or rotational frequency that coincides with its natural frequency. Operation at critical speed can cause destructive shaft vibrations. The critical speed margin is the rpm spread between operating speed and critical speed. Existing shafts were shorter and generally operated at lower speeds, see Table 1. The problem, therefore, was to design a shaft that met all the requirements of speed/torque and misalignment capability and could still operate with a safe critical speed margin.

Design Requirements

It was quite obvious at the start that an "off-the-shelf" PTO shaft was not available to meet the requirements of the FSW aircraft. Therefore, the initial step was to establish the design requirements for an acceptable drive shaft. A design specification was prepared using military specification MIL-S-7470[1] as a basis. A design layout was completed to establish the envelope restrictions such as length and diameter and to establish the drive shaft interfaces. The shaft operating speed range was determined by applying the engine output shaft drive ration to the engine operating speed (1/1 based on N_2 rpm). The static strength requirements of the shaft was set at twice the maximum possible starter torque to which the shaft could be subjected. Shock loading was not a consideration since the jet fuel starter (JFS) drive system incorporates a torque converter disk clutch.

Maximum torque when the engine is driving is much less than starter torque. To protect the engine from any possible overtorque condition, the shaft was required to incorporate a shear section. This shear section is identical to the one in the F-18 PTO shaft since the F404 engine is common to both airplanes. Table 2 presents the various torque values that were important considerations in the shaft design.

The X-29A aircraft was judged to have a very flexible structure and much relative motion was expected between the engine drive and the AMAD input shaft. An important design parameter was to establish the shaft requirements for both angular and axial misalignments. A dynamic study was performed to determine the maximum possible flexing motion during various aircraft maneuvers. In addition to *g* loading, thermal growth, manufacturing tolerances of significant components, and assembly and installation tolerances were also considered. While establishing the shaft misalignment requirements, shaft insertion position also had to be determined. The AMAD/PTO shaft interface has a sliding spline with approximately ½ in. axial play. This is common with the F-16 AMAD design. The initial insertion position is extremely important because the distribution of the ½ in. play on either side of the initial installation position has to be selected such that during flight the forward end of the PTO shaft does not bottom out inside the AMAD gearbox or that spline engagement does not fall below a safe minimum if the withdrawal movement is too far rearward. Basic design features and requirements for the X-29 PTO shaft are presented in Table 3.

Initial Design

The Bendix Fluid Power Division, an experienced supplier of aircraft PTO shafts, was selected to design and build the X-29A shaft. Various metal shaft designs using "off-the-shelf" components were evaluated. Critical speed analyses were performed with various shaft configurations and materials. Table 4 presents tabulation of the relevant material properties of the basic three materials considered in this study — aluminum, titanium, and graphite/epoxy composite. None were found that could provide sufficient critical speed margin above the PTO shaft operating speed. Increasing the shaft diameter to increase the shaft stiffness was not successful, since this also increased the shaft weight. Bendix's efforts to study the PTO shaft dynamics were supported by Grumman, General Electric (the engine manufacturer), and Sundstrand (the AMAD supplier). Close industry coordination led to the common realization that, in addition to shaft considerations, the shaft end conditions (i.e., the bearing stiffnesses of the end supporting gearboxes) were also extremely important.[2] The

Presented as Paper 85-4045 at the AIAA/AHS/ASEE Aircraft Design Systems and Operations Meeting, Colorado Springs, CO, Oct. 14–16, 1985; received Nov. 11, 1985; revision received Aug. 18, 1986.

*Power Plant Design Group Leader.

stiffnesses of the engine gearbox and AMAD gearbox were reevaluated. Actual tests were performed to measure their stiffnesses. It was discovered that the AMAD stiffness could be increased significantly by installing a stiffening collar between the "snout" (AMAD PTO shaft drive housing) and a nearby hydraulic pump pad (Fig. 3). This rework was accomplished by Sundstrand. The new stiffness test value of 78,000 lb/in. was significantly higher that the previous value of 34,000 lb/in. During this effort, Bendix recommended that a composite shaft be considered because of its low weight and high stiffness. After evaluation and testing, the composite shaft was successfully used as the center shaft in the Bendix PTO shaft assembly.

Composite Shaft Design

The composite shaft provides reduced weight and increased stiffness, which are major factors for increasing critical speed. See Fig. 4 and Table 5. This shaft design utilizes a thin-wall carbon filament wound (six layers) torque tube with titanium end flanged attaching fittings. The carbon filaments were wound at alternate 14-deg angles to the longitudinal shift axis. The angle of the winding was optimized to meet speed and torque requirements. A patented, no-bolt/pin-positive mechanical joint provides the transition between the composite tube and titanium end flanged fittings.

Complete Shaft Assembly

The composite shaft provides only the rigid center section of the complete PTO shaft assembly (see Fig. 5). Flexible coupling end fittings are added at each end to form the complete assembly. The end flexible assemblies (Figs. 6 and 7) incorporate two diaphragms each with a ball joint at the center of articulation. The diaphragms provide the angular shaft flexibility. The ball joints are necessary to absorb any axial load that the shaft might experience, since the diaphragms can tolerate very little axial tension. Any axial misalignment is relieved by the sliding spline interface between the AMAD and the PTO shaft. The frictional force to slide the splines varies with the load being transmitted by the splines, but should not exceed 225 lb. The ball joints also provide a safety "antiwhipping" feature in the shaft design. In the event of bellows failure, the relative positions of the PTO shaft sections on each side of the failed bellows are maintained by a centering ball arrangement at the center of the bellows. This feature precludes shaft "whipping" with a failed bellows. However, no rotary power is transmitted by the centering ball. The end assemblies also feature a three bolt flange for attachment to the center composite shaft. The AMAD end assembly features a spline end to mate with the spline drive in the AMAD housing. The engine end assembly incorporated a shear section (bolts) and a flange for attachment to the engine PTO drive. This flange features three attaching bolts and a shield to protect the diaphragms from wrench damage.

Critical Speed Analyses

During the shaft design phase, extensive critical speed analyses were performed in view of the shaft length and high

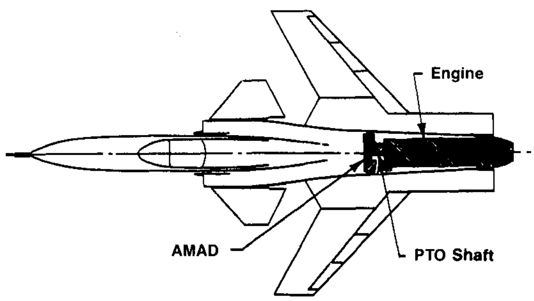

Fig. 1 X-29A aircraft.

Table 1 PTO shaft comparison

Aircraft	Engine	Shaft speed, rpm	Shaft length, in.
X-29A	F404	16,810	31.52
F-16	F100	14,643	23.05
F-18	F404	16,810	17.14
F-14A	F101 DFE	15,354	9.00

Table 2 X-29A PTO shaft torque design requirements, lb-in.

Engine driving, max	700
Starter peak, max	4250
Engine requirement, shear section	4800–5400
Static limit torque	8500

Table 3 Design features and requirements of C-29 PTO

Connects engine to AMAD
F-18 GE F404 engine
F-16 Sundstand AMAD
AMAD aircraft mounted
AMAD accessories face aft
Engine PTO pad 17.7-in. aft of inlet
31.52-in. PTO shaft required
Rate conditions
  Power 100 hp
  Torque 420 lb-in.
  Speed 15,000 rpm
Maximum conditions
  Power 185 hp
  Torque 700 lb-in. (engine driving)
    4250 lb-in. (starter peak)
  Speed 17,190 rpm
Static limit torque: 8500 lb-in.
Deflection capability 1.5-deg max each end
Critical speed: 20,000 rpm (min)

Table 4 PTO shaft materials properties

Material	Strength tensile ultimate, psi	Stiffness Youngs modulus, psi $\times 10^6$	Density, lb/in.^{3}
Aluminum	42,000	10.0	0.100
Titanium	79,000	16.0	0.160
Graphite/epoxy composite	22,000	14.2	0.059

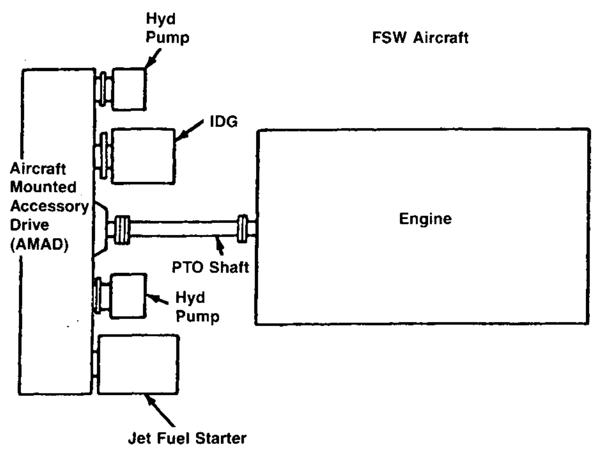

Fig. 2 General arrangement of aircraft drive mechanism.

Table 5 Design features of graphite/epoxy center shaft

Graphite: Toray T300B
Resin: LY556 Araldite
Filament wound at ±14-deg angle
Glass tape outer cover
Design features
High torque
High modulus
Low weight
Positive mechanical joint
Excellent environmental resistance
Composite structure allows design flexibilty. Angle of filaments selected to meet torque and speed requirements

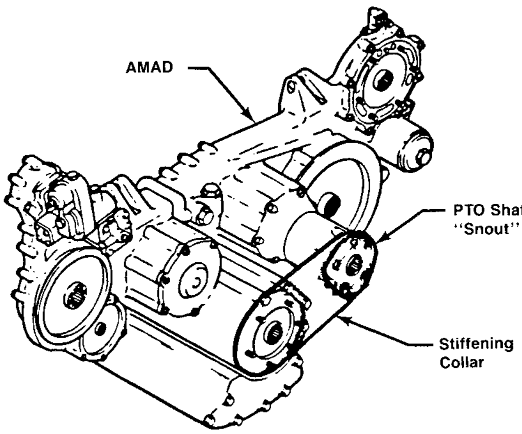

Fig. 3 AMAD stiffening collar.

speeds. These analyses utilized finite-element models of the entire rotating subsystem, which included the PTO shaft and the engine and remote gearbox drive shafts. Initial analyses demonstrated that the existing shaft designs utilizing available metal materials would not work. The critical speeds calculated were below the operating speeds. When the composite shaft was introduced, analysis showed a critical speed of only 18,340 rpm, which was above operating speed, but the positive margin was small. To increase this margin, a shaft weight-reduction program was instituted. Some weight was removed from the titanium end flanges of the composite shaft. Additional weight was saved when Bendix introduced a bimetallic joint at the AMAD end flexible assembly. The heavier all-steel construction was modified so that steel was used only for the male splines where greater wear resistance was required. However, titanium replaced the steel material from the bimetallic joint through the diaphragms and the flange. This can be seen in Fig. 7. Thus, the weight of the complete shaft assembly was reduced from a calculated 7.1 to a 5.9 lb. Increasing the AMAD stiffness and using the composite shaft design resulted in a projected critical speed of over 20,000 rpm. Since the maximum steady-state operating shaft speed is 17,190 rpm, this value of critical speed was judged to be satisfactory, providing a margin of over 20%. The spring rates of the components in the PTO shaft subsystem are presented in Fig. 8. Subsequent testing verified adequate critical speed margin.

Support-Shaft Concept

During the single-shaft design effort some consideration was also given to a dual- or supported-shaft design. This concept reduced the unsupported span between the engine drive and the AMAD "snout" by adding a support post from the aircraft structure. Replacing the one long shaft with two smaller ones, however, created many design problems. These included increased complexity of the shaft assembly as well as adding additional structure to the aircraft. A support post with high modulus would have to be installed in the aircraft to

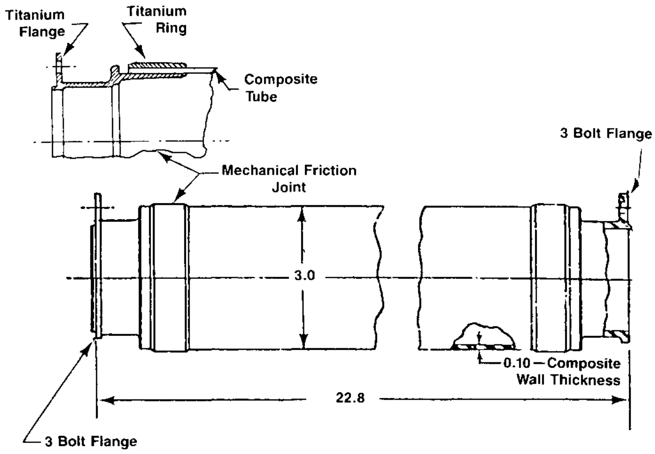

Fig. 4 Composite center shaft.

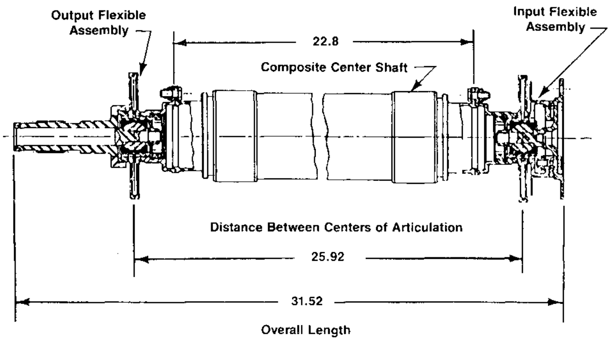

Fig. 5 PTO shaft assembly.

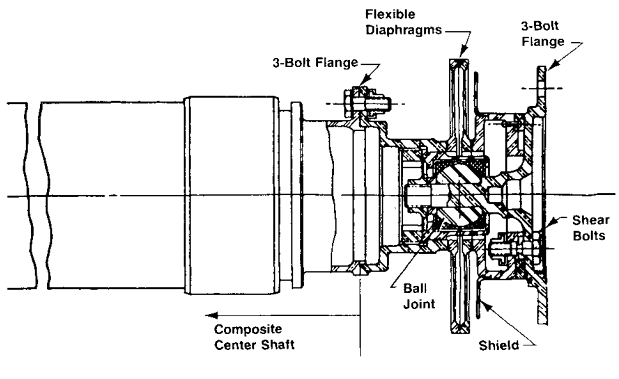

Fig. 6 Input coupling assembly (engine end).

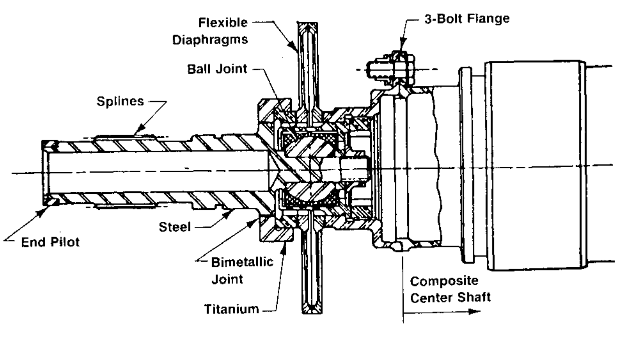

Fig. 7 Input coupling assembly (AMAD end).

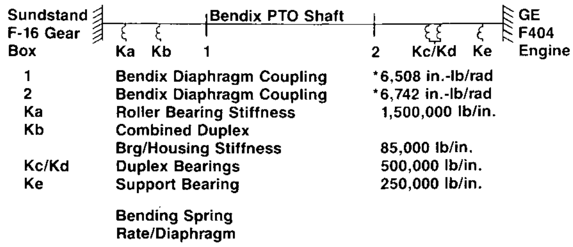

Fig. 8 X-29A single-composite shaft critical speed analysis spring rates.

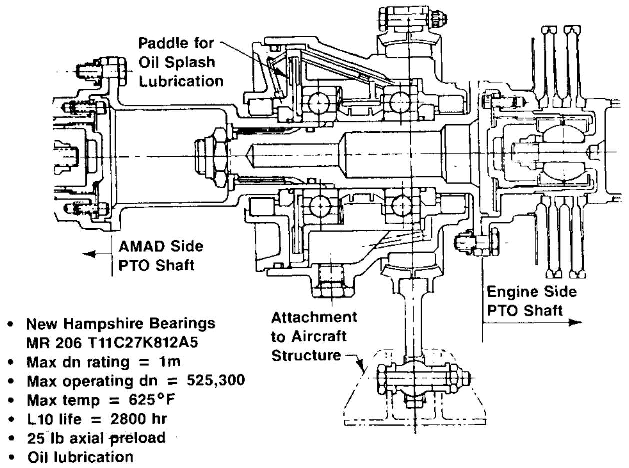

Fig. 9 Dual shaft-hanger bearing box.

support a high-speed bearing. The bearing would require cooling because of its speed and loading, thus requiring and oil cooling system. The hanger bearing box design is shown in Fig. 9. The reduced span between the engine and the support bearing attached to the aircraft post would reduce the capability of the PTO shaft to handle offset misalignments (Fig. 10). Thus, the diaphragm assembly of the PTO shaft would have to be designed for greater misalignment capability. This would have a negative effect on the shaft critical speed. The supported shaft design was completed, but never built or tested.

Testing

Extensive testing was performed to establish design and manufacturing adequacy of the PTO shaft. Starting with material and component testing through assembly and subsystem level testing, the ability of the shaft to meet its design requirements was proved. Torque tests to failure demonstrated a strength of almost seven times maximum operating torque. Critical speed tests demonstrated successful operation at speeds of over 20,000 rpm. Subsystem level testing performed by Sundstrand included the three rotating shafts in the subsystem, the engine gearbox output shaft, the PTO shaft, and the AMAD shaft. Although each of the three shafts was individually balanced by the manufacturer, some additional balancing at the subsystem level was required. The three shafts were indexed to various positions for minimum runout and, where necessary, balance weights were added to the complete rotating system.

Composite Center Shaft Testing

The following tests were performed on the composite material and composite center shaft without flexible end assemblies:

1) Materials properties testing of shaft cutoffs at −40 to +250°F.

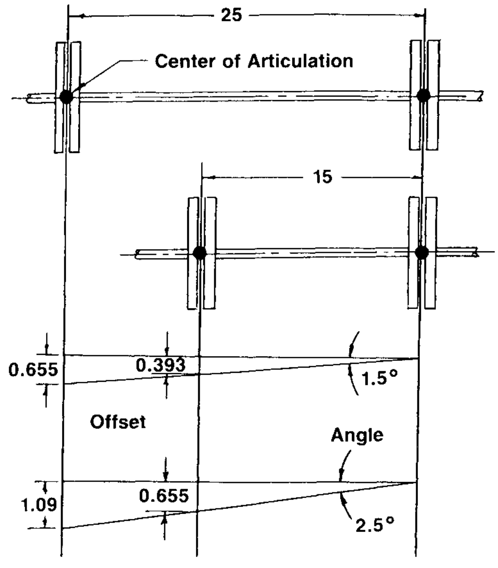

Fig. 10 Misalignment capability vs shaft length.

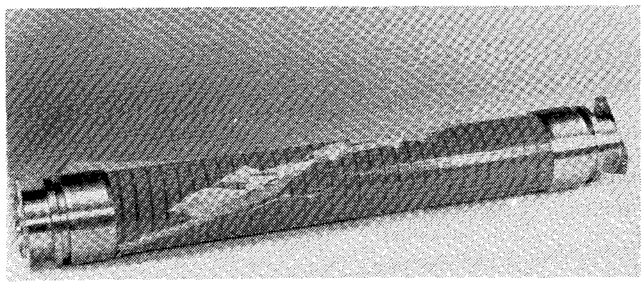

Fig. 11 Torque failure shaft.

2) Destructive torque testing of three complete shafts made from same batch of material. Two reached limit of test setup without failure (over 30,000 lb-in. torque). One shaft failed at 29,100 lb-in. torque in a spiral 14-deg angle separation (Fig. 11) along entire length of the composite section due to fiber rupture at a shearing stress of 22,766 psi (predicted ultimate was 22,000 psi).

3) Hoop strength testing to simulate centrifugal forces encountered at a shaft speed of 17,000 rpm with a safety factor of 1.5. The limit pressure test at 100 psi and burst at 420 psi.

4) Fatigue test—6.68×10^6 cycles of 425 to 4250 in.-lb torque.

5) Ultrasonic inspection of the production shafts and the fatigue shaft before and after fatigue testing showed no discrepancies.

6) In the fast Fourier transform (FFT) vibration analysis, the natural frequency was 560 Hz, equivalent to 33,660 rpm.

Qualification Testing of the Complete Shaft Assembly

The following tests were performed:

1) Vibration analysis (FFT): natural frequency was 350 Hz, equivalent to 21,000 rpm.

2) Dynamic balance (0.02 oz.-in.).

3) Critical speed to 20,000 rpm.

4) Overspeed to 21,000 rpm.

5) Endurance: 100 h at 700 lb-in. torque with shaft misaligned 1 deg and speeds up to 17,520 rpm.

6) Fatigue torque: 10×10^6 cycles, 100–1000 lb-in. torque in a −40 to +250°F temperature environment.

7) Humidity.

8) Salt fog.

9) Temperature shock.

10) Vibration.

11) Fluid deterioration test with the following fluids: oils MIL-L-23699 and MIL-L-7808, hydraulic fluid MIL-H-83282, and fuel MIL-T-5624.

Subsystem Testing

Subsystem level testing was performed with the actual flight hardware, including the engine gearbox, PTO shaft, and AMAD.[3] The shaft instrumentation included six proximity probes on the PTO shaft to measure lateral deflections and four accelerometers mounted on the gearbox housings at either end of the shaft to measure vibratory *g* loads. See Fig. 12. In addition, a photosensitive pickup was used to generate a 1 pulse/rev. This pulse was used to trigger a Hewlett-Packard 5420A digital analyzer for phase relationship data reduction.

The three rotating shafts and the end support bearings were tested to determine the critical speed of the system. The initial results were disappointing—shaft deflections were excessive within the operating speed range. The testing was initially performed at zero misalignment. Subsequent testing with the shaft at maximum design misalignment (1.5 deg) showed no measurable effect on the vibration or shaft deflections. During this testing, other parameters were altered to determine their effect on the shaft displacement. Increasing the gearbox accessory loading from 20 to 80 hp decreased shaft deflections slightly at high speeds. Apparently, an increase in the horsepower loading of the accessories caused an increase in bearing loads and, therefore, an increase in bearing stiffness. Shaft system runout and balance were found to be very significant in obtaining low deflections and high operating speeds. During testing, it was noticed that as the shaft was slowly rotated the air gap between the shaft surface and the proximity probe varied considerably. This runout condition was minimized by selective indexing of the PTO shaft and the engine and AMAD shafts to various positions at their interfaces. Improved balancing of the rotating system was then performed. The shaft deflections and their phase angle relationship were measured. A corrective weight and location was calculated and applied in the form of a washer at the engine to PTO shaft flange. Using this field balancing procedure, acceptable shaft deflections (0.005-in. maximum double amplitude) were obtained through shaft speeds of up to 20,000 rpm. This testing demonstrated that, for the three shafts tested, lateral deflections were closely related to the gearbox housing vibratory loads, see Figs. 13 and 14.

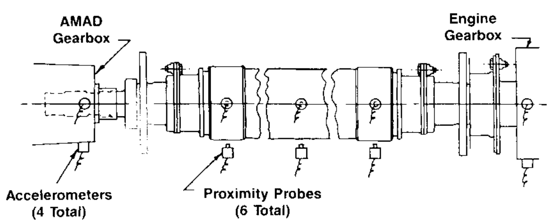

Fig. 12 PTO shaft subsystem instrumentation.

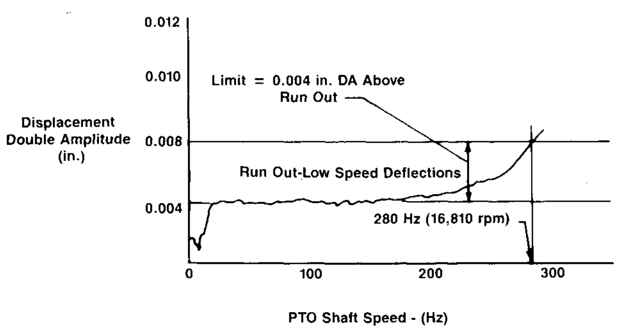

Fig. 13 PTO shaft vibration survey signature profile deflections.

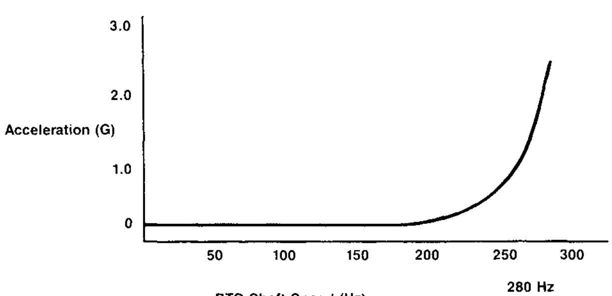

Fig. 14 PTO shaft vibration survey signature profile accelerations.

Aircraft Testing

After aircraft installation, the PTO shaft was instrumented with proximity probes and the gearbox housings with accelerometers. During ground engine testing, measurements were taken of the shaft deflections and vibrations. The low values measured agreed closely with the subsystem level test results. For flight monitoring of the PTO shaft gearbox, accelerometers were the only instrumentation utilized. The acceleration vs speed (*g* vs rpm) signature profile curve from ground test data was used as a guide in evaluating flight performance. The PTO shaft has performed satisfactory during flight testing of the X-29 aircraft. No problems have been experienced.

Conclusion

The power takeoff shaft required for the X-29A experimental aircraft was successfully obtained by the practical application of good design techniques, the establishment of clear design requirements, and the use of aerospace industry expertise to assist in the design solution. An industry/product survey was conducted to find the most suitable product/material to satisfy design requirements. Utilization of adequate testing was made to confirm design solution.

The following lessons have been learned:

1) Designing for the minimum runout and small dynamic imbalance that is normally taken for granted in a simple rotating shaft must not be overlooked when designing a more complex rotating subsystem. This can easily be overlooked when the individual parts are designed separately and then subsequently assembled into a subsystem.

2) To obtain the desirable low vibration in the shaft operating speed range, the shaft must be designed for a critical speed margin of 20% minimum.

3) Not only are low shaft weight and high shaft stiffness desirable in high-speed shaft design, but high stiffness must be achieved in the end support gearboxes.

References

[1]Shaft, Power Transmission, Aircraft Accessory, General Specification for Military Specification MIL-S-7470A, Dec. 9, 1975.

[2]Ferguson, J.H. and Stocco, J.A., "Avoiding Destructive Shaft Vibrations, Machine Design, Allied Corp., Bendix Fluid Power Div., Utica, NY, June 1984.

[3]Johnson, H.F. and Maresko, W.A., "Critical Speed Testing of the Grumman X-29A PTO Shaft Subsystem," SAE Paper 841603, Oct. 1984.